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# SSME BLADE DAMPER TECHNOLOGY

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Before 1975 turbine blade damper designs were based on experience and very simple mathematical models. Failure of the dampers to perform as expected showed the need to gain a better understanding of the physical mechanism of friction dampers. Over the last 10 years research on friction dampers for aeronautical propulsion systems has resulted in methods to optimize damper designs.

The first-stage turbine blades on the SSME high-pressure oxygen pump have experienced cracking problems due to excessive vibration. A solution is to incorporate a well-designed friction dampers to attenuate blade vibration. The subject study, a cooperative effort between NASA Lewis and Carnegie-Mellon University, represents an application of recently developed friction damper technology to the SSME high-pressure oxygen turbopump.

The major emphasis of study was the contractor's design known as the two-piece damper. Damping occurs at the frictional interface between the top half of the damper and the underside of the platforms of the adjacent blades. The lower half of the damper is an air seal to retard airflow in the volume between blade necks.

A bench test apparatus was developed to conduct an extensive set of experiments on the two-piece damper. The bench test apparatus was successful. The normal load was applied through a fishhook-pulley-weights system. The weights were varied from 0.25 to 70 lb. The blade support was tuned to simulate blade natural frequencies at pump operating temperatures and speeds (approximately 9.5 KHz for the edgewise bending mode). The excitation system consisted of an electromagnet and a small chip of transformer iron mounted on the blade. Blade response, measured with a miniature accelerometer, was up to 700 G's tip acceleration, 0.1-mil tip displacement, and 1700-psi stress at the crack location.

Analytical models were fit to the experimental data and used to extrapolate the results to pump operating conditions. These extrapolations show that the best thickness for the two-piece damper is 0.047 in. and that the performance can be improved by reducing the width by 15 percent. Even with these improvements, the performance is poor. An example of predicted damper performance (pump conditions) is attached. The linear portion of the curve, emanating from the origin, is where the damper is locked and not functioning. The curved portion, beginning at an input level of 3 lbf, is where the damper starts to slip. The optimum condition is when the damper slips for approximately one-half of the vibratory cycle. Since the stress required to crack the blade is estimated to be between 5000 and 10000 psi, we conclude that the damper is not working in the range of interest and is a poor design.

Two other damper designs were considered. A design, known as the X-damper, was found to be a poorer performer than the two-piece damper. A tip damper design was found to have excellent performance.

Whirligig tests are inconclusive concerning two-piece damper performance. However, these results strongly imply that the parameter controlling blade vibrations is tip clearance. Large tip clearance results in high stresses.

The hot-fire test results from MSFC are the only one which imply that the two-piece damper is effective in reducing stresses. Although this may be the case, the lack of cracked blades may be due to something other than the two piece dampers. For example, if the tip clearances on the blades in the hot-fire test are relatively tight, the blades would have low stresses without dampers.

In summary, the two-piece and X-dampers are poor performers. The tip damper is a good performer and should be more actively pursued. Because of the unknowns associated with tip clearance, the hot-fire test should not be used as a justification to incorporate the two piece damper into production. Additional testing should be conducted to understand the role of tip clearance on blade response.

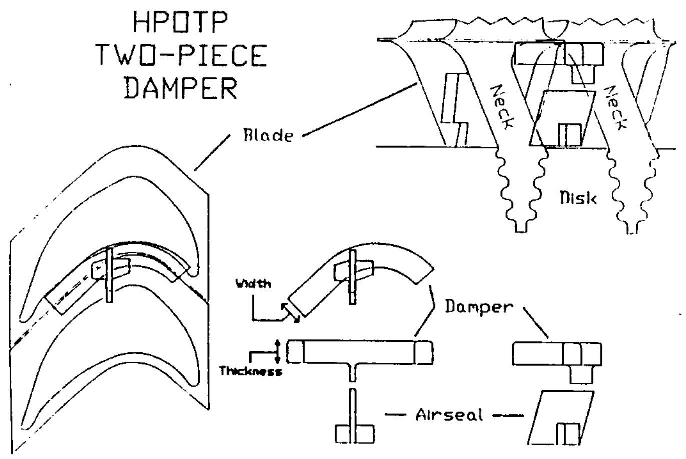


Figure 1.

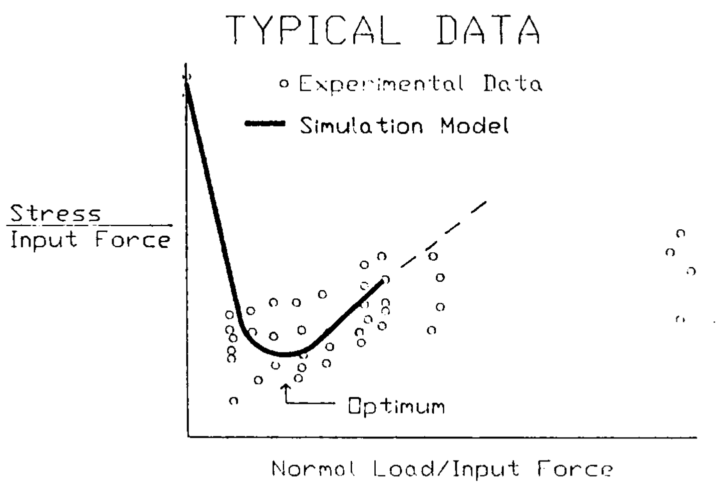


Figure 2.

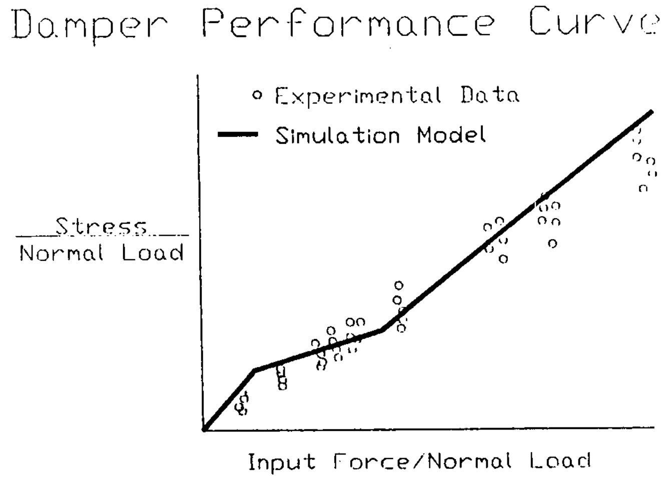


Figure 3.

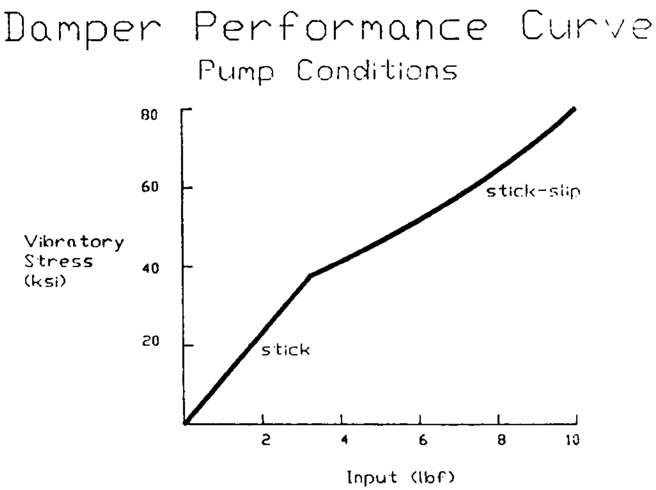


Figure 4.